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Study of the Behaviour of an Exhaust Gas Heat Exchanger for Improved Engine Warm-up

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Abstract: Because of the many short distance journeys undertaken by vehicles, engine cold-start is a target for significant fuel consumption benefits. By reducing the warm-up time, engine friction losses can be reduced and cabin heating can be provided sooner. This can be achieved by increasing the available thermal energy per unit thermal mass. In this paper the effects of a heat exchanger are investigated to use otherwise waste heat from exhaust gases. The device was installed in the coolant circuit of a 2.4L Diesel engine and experiments were conducted over 25°C-start New European Drive cycle. The system offered a small oil and coolant warm-up benefit of 1-4°C over phase 1 of the drive cycle. However, this improved warm-up was not reflected in benefits in fuel consumption. The limited impact on engine performance was a result of the increased thermal inertia required to integrate the heat exchanger into the system. The benefits are strongly dependent on engine duty cycle and it was estimated that to achieve a significant improvement in oil warm-up rate an additional thermal mass of less than 1kg water should be the design target.

Keywords: Thermal Management, Waste Heat Recovery, Fuel consumption

1. Introduction

Engine fuel economy and emissions from cold-start are significantly worse than under fully warm conditions because of increased frictional losses and colder combustion temperatures. Although conventional thermal management systems have been aimed at providing sufficient engine cooling, these systems have a key role to play in optimising engine warm-up. Recently, these systems have been developed for improved warm-up through the use of active components such as flow control valves, split cooling circuits and clutched or electric water pumps. However, studies from cold-start suggest that engine warm-up can only be improved by increasing the *available heat energy per unit thermal mass*. This can be achieved either by reducing the thermal inertia or by increasing the total available heat energy. This paper focuses on the latter of these two approaches by using otherwise waste heat from exhaust gases to increase thermal energy available during warm-up.

2. Background

Improving engine warm-up rate from cold-start has a number of benefits both in terms of performance and comfort. Higher oil temperature reduces engine friction through changes in oil viscosity and the availability of free excess heat provides for good cabin heating. However, following cold-start the thermal demands are high and the available energy is scarce. Therefore, design changes that reduce this energy demand or increase the available energy will be beneficial to product performance [1,2]. This is achieved either by reductions in overall thermal inertia or by increasing the available thermal energy during warm-up.

The first of these two approaches can be achieved by reducing the overall mass of the engine and thermal management system through design optimisation. Alternatively a number of concepts aim to reduce the thermal inertia temporarily during warm-up by isolating parts of the oil and coolant circuits using flow control valves, electric or clutched water pumps and novel oil sump designs [3-6]. These approaches reduce the thermal mass participating in the warm-up, thus reducing the time to reach operating temperature.

The second approach involves an additional heat source to provide additional thermal energy during warm-up. From a fuel consumption perspective, it is essential that this additional heat be from a *free* source, as concepts that involve additional energy usage such as fired cabin heaters will be detrimental to fuel economy [7]. Otherwise waste heat can be sourced from the coolant circuit which may exhibit losses to ambient [3], from the exhaust gases [8] or through storage from a previous warm operation [9-11].

Kunze et al. [12] modelled the effect of external heat addition and predicted a 1.5% reduction in fuel consumption following a 2MJ heat addition over cold-start NEDC. They did not offer any suggestions for practical implementations, and experimental work where engine oil was preheated before the experiment did not confirm the results from their model. The work by Andrews et al. [8] is a practical implementation of the study by Kunze et al. Heat was added to the coolant during the warm-up period by installing a coolant heat exchanger in the exhaust manifold. The heat was then transferred to oil via another heat exchanger. The tests were run from cold-start on a steady state rig. The additional heat increased the warm-up rate of the oil by an average of 8 to 12^{o}C, yielding a 12-15% reduction in instantaneous fuel consumption. These show that waste energy recovery has a large potential for improved warm-up, however designing a system that is subsequently capable of rejecting the excess heat under fully warm conditions is problematic [13]. On modern engines, such a system needs to consider exhaust gas after-treatment that also require exhaust gas heat from cold-start.

This work presented here is part of a larger project concerned with engine thermal management. An active thermal management system had been designed and installed on a production, EURO IV emissions specification, 2.4L Diesel engine. This active system included a number of flow control valves to control coolant flows in different legs of the circuit, including coolant flow through the oil/coolant heat exchanger. In addition a dual EGR cooler system was installed whereby EGR gases could be cooled using coolant or lubricant. Further details on the behaviour of this system can be found in previous publications [2,3]. The conclusions from this work were that warm-up could be improved by a temporary reduction in thermal inertia from isolating the front end coolant circuit. Control of heat flows from EGR gases to the coolant or lubricant created a trade-off between the upper and lower engine warm-up, controlled by the two fluids respectively. This work builds upon these findings and focuses on the behaviour of an exhaust gas heat exchanger integrated into this prototype system.

3. Experimental Approach

3.1 Experimental Facilities

All experiments were carried out on the engine dynamometer facilities at the University of Bath. The 2.4L, EURO IV emissions specification turbocharger Diesel engine was installed on a transient engine dynamometer. Engine cooling was provided using a fan replicating the air flow over the radiator as a function of simulated vehicle speed. In each case, cold-start experiments were carried out from a 25^{o}C start temperature and consisted of an emulated New European Drive Cycle (NEDC). The drive cycle was defined such as to replicate a light commercial vehicle and the speed and torque trace are shown in figure 1.

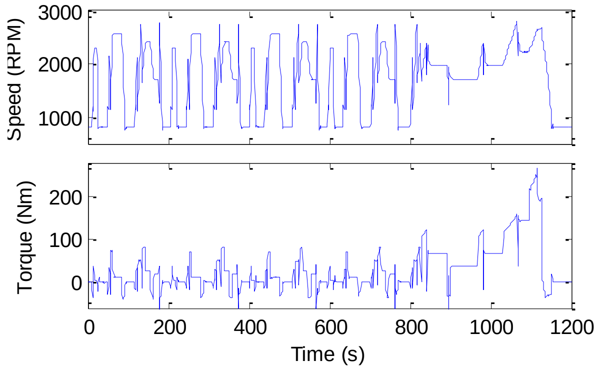

Figure 1: Speed and torque trace of emulated drive cycle

Fuel consumption was measured using both a gravimetric fuel balance and by carbon balance of the feed-gas emissions using appropriate correction procedures to improve the measurement accuracy [14]. Emissions concentrations were measured using Horiba MEXA 7000 emissions analysers. Over 100 thermocouples were installed to measure the metal and fluid temperatures both within the engine structure and around the internal and external circuits. A number of these thermocouples were arranged in arrays of three to create multi-point sensors, allowing thermal gradients to be measured and the local heat flux to be calculated [15].

3.2 Heat Exchanger Hardware

The exhaust gas heat exchanger used in this work was taken from a production, medium-sized passenger car. In that application it was used to improve cabin heating performance from cold-start. The exhaust gas heat exchanger layout is shown schematically in figure 2 and a photograph is shown in figure 3. The heat exchanger itself is a simple gas to liquid counter-flow device, although some complexity arises from its installation within the engine system. The primary purpose of the heat exchanger is to allow waste heat from the exhaust gas to be captured by the coolant to assist with warm-up. Once the engine is warm, the heat exchanger is unnecessary and indeed counter-productive. In order to avoid undesirable heat gain which must then be rejected by the radiator and also to avoid boiling the coolant flowing through the heat exchanger when the engine has warmed up, it is necessary to divert the gas flow. Diverting or stopping the coolant flow will always leave stagnant coolant within the heat exchanger which will boil as hot exhaust gases pass through the device. To this end, the device is equipped with an integral bypass leg and vacuum operated actuator controlling a butterfly valve. A simple control algorithm was implemented using *Accurate Technologies ATI vision 'No hooks'* which switched the gas flow to bypass mode when a critical coolant temperature exiting the heat exchanger was reached.

The heat exchanger effectiveness was calculated from temperature measurements of the exhaust gas and coolant at inlet and outlet to the device according to equation 1.

$$\varepsilon = \frac{q}{q_{\max}} = \frac{C_h\left(T_{h,i} - T_{h,o}\right)}{C_{\min}\left(T_{h,i} - T_{c,i}\right)} \quad [1]$$

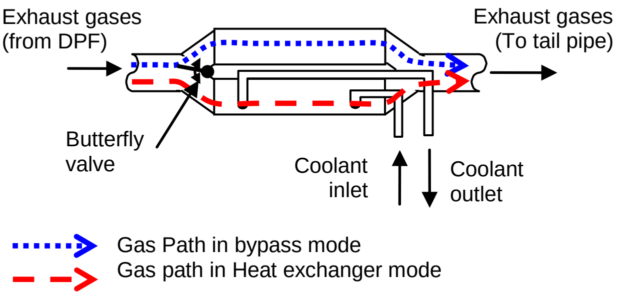

Figure 2: Exhaust gas heat exchanger layout and operation

Figure 3: Exhaust gas heat exchanger

3.3 Heat Exchanger Installation

A number of practical aspects were taken into account when installing the exhaust gas heat exchanger both on the gas side and the liquid side. It was decided that a near production setup should be adopted, within the constraints of the experimental facility. The gas side offers significantly less flexibility because of emissions regulations on exhaust gases. To meet EURO VI emissions standards, it is almost certain that the exhaust will be equipped with a catalyst and particulate filter (DPF). The regeneration and light off requirements mean that an exhaust gas heat exchanger would need to be installed downstream of these devices. The engine did not include a DPF because of issues of regeneration and filter state that could affect testing repeatability so to best replicate this device, the exhaust gas heat exchanger was installed several meters downstream from the catalyst. Not only would this affect gas side temperature at the heat exchanger, but would also give realistic hose lengths for the liquid side.

For the liquid side, the heat exchanger was integrated into a prototype active thermal management system previously developed during this project [3]. This system comprised of an engine-out coolant throttle, a second throttle in the EGR cooler leg and a dual EGR system whereby EGR gases may be cooled either by coolant or lubricant. The layout of this circuit is best understood by considering figure 5 without the exhaust gas heat exchanger and for a detailed analysis of the behaviour of this system the reader is directed to our previous publication [3]. Under fully warm conditions the coolant temperature is controlled passively through a wax element pressure regulated thermostat (PRT) [16]. Two experimental setups were considered corresponding to two experimental phases and in both cases the additional coolant volume was approximately 2L:

- Heat exchanger installed in an independent circuit
- Heat exchanger integrated into main cooling circuit

Figure 4 shows the heat exchanger installation in an independent circuit. This circuit links the exhaust gas heat exchanger to the oil cooler to provide additional heat to the oil during warm-up. Coolant flow was provided from an electric pump with variable speed control providing flows up to 10L/min. Clearly this configuration does not allow cooling of oil under fully warm conditions, but is sufficient for this prototype research project. In this configuration, the controllers for the main coolant circuit were setup to their previously optimised setup and a range of different secondary circuit coolant flow rates were tested using the variable speed pump.

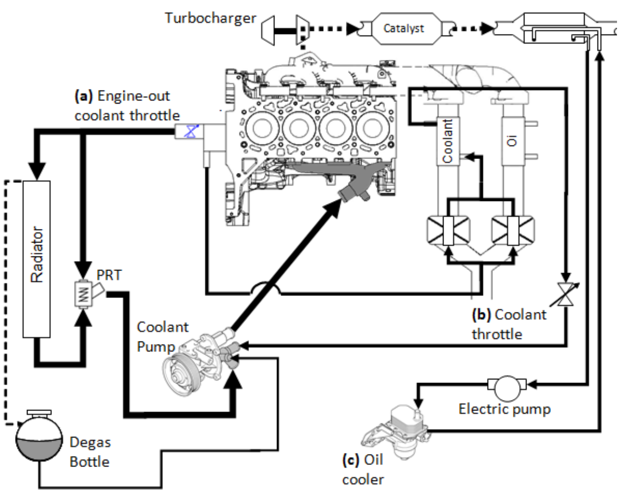

Figure 4: Exhaust gas heat exchanger included as secondary circuit

Figure 5 shows the exhaust gas heat exchanger integrated into the main coolant circuit. The circuit again provides heat to engine oil through the oil cooler, but additionally the heat from the exhaust can improve coolant warm-up. In this configuration, three different control approaches are possible with different aims:

1. All heat to oil (using oil-cooled EGR and oi-cooler)
2. All heat to coolant (using coolant-cooled EGR and bypassing oil cooler)
3. A intermediate of the two previous aims

A design of experiments (DoE) approach was adopted to understand the effects and interactions of three of the circuit actuators:

1. Coolant throttle in EGR/Exhaust gas heat exchanger/oil cooler circuit (***b*** in figure 5)
2. Oil cooler control valve (on or bypass)
3. EGR cooler type (coolant or oil)

In all cases, the engine out coolant throttle (***a*** in figure 5) remained closed during warm-up. The heat exchanger butterfly valve was switched to bypass mode only when the coolant temperature reached 90^{o}C. The test programme is detailed in table 1.

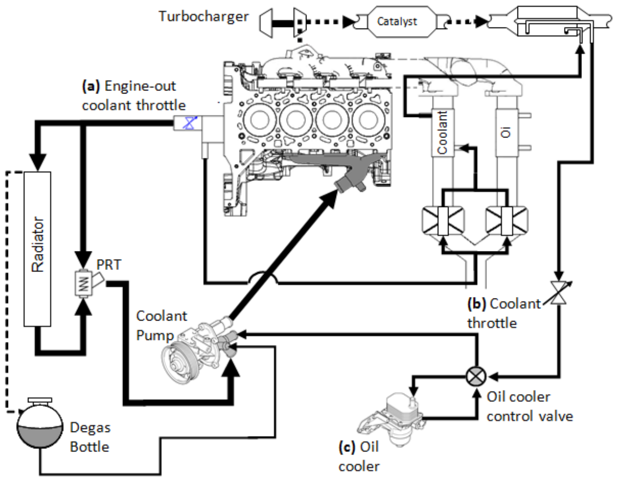

Figure 5: Exhaust gas heat exchanger integrated into main coolant circuit

DoE No.	EGR/HX/oil cooler flow at idle (L/min)	Oil cooler bypass valve	EGR cooler type
1	0.8	On	Coolant
2	3	On	Coolant
3	6	On	Coolant
4	0.8	On	Oil
5	3	On	Oil
6	6	On	Oil
7	0.8	Bypass	Coolant
8	3	Bypass	Coolant
9	6	Bypass	Coolant
10	0.8	Bypass	Oil
11	3	Bypass	Oil
12	6	Bypass	Oil

Table 1: DoE test programme for integrated circuit assessment

Analysis of the results from the DoE approach was performed using simple response models fitted to the experimental data. In each case these were simple polynomial models characterising the thermal behaviour, fuel consumption and emissions. The general model structure is described in figure 6 and the quantification of various measurements over the drive cycle is detailed in table 2.

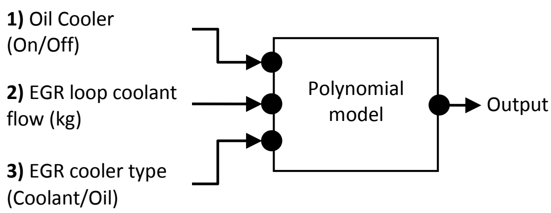

Figure 6: General model structure for DoE results

Measurement	Quantification
Temperature	Temperature rise over transient cycle (^{o}C)
Coolant Flow	Cumulative flow over transient cycle (kg)
Fuel consumption/ Emissions	Cumulative use/production over transient cycle (g)
EGR cooler/ oil cooler control	Qualitative variables – Represented as -1 or 1 in modelling structure

Table 2: Quantification for transient measurements for DoE response models

3.4 Analytical analysis

Following on from the two experimental campaigns, a semi-analytical assessment of the new system was conducted. Initially the measured heat exchange in the exhaust gas heat exchanger was used to calculate the time wise temperature rise of the additional thermal mass resulting from the heat exchanger installation according to equation 2. Only the coolant and hoses were considered in the calculation of the system thermal capacity.

$$dT_{cool} = \frac{q_{HX}}{\sum m_i C_i} \qquad [2]$$

In a second phase, the heat transfer to oil was simulated using Newton's law of cooling and a calculated heat transfer coefficient for the oil cooler. In this case the coolant temperature was calculated according to equation 3.

$$dT_{cool} = \frac{q_{HX} - (hA)_{oilcooler}(T_{oil} - T_{cool})}{\sum m_i C_i} \qquad [3]$$

These two simple approaches give an estimate of the effect of additional thermal inertia on the performance of the exhaust gas heat exchanger system.

4. Results

4.1 Heat exchanger performance

The behaviour of the exhaust gas heat exchanger was characterised using the secondary circuit installation. Figure 7 shows the heat exchanger effectiveness calculated according to equation 1 from both gas and coolant temperatures. The results are very different, however the small difference between inlet and outlet coolant temperatures mean this measure is ill-conditioned and not expected to be an accurate measure of effectiveness, notably just after cold-start. Based on gas side measurements, over the first 700 seconds of the drive cycle the effectiveness remains high, varying between 0.65 and 0.95 with a tendency to drop as the drive cycle progresses. This can be explained by the increase in metal temperature of the heat exchanger which reduces the thermal gradient between the gases and the metal, reducing convective heat transfer. Over the following 150 seconds, there is a transition period as the coolant temperature fluctuates around 90°C and the simple on/off control algorithm successively engages and bypasses the heat exchanger on the gas side. After 850 seconds the bypass valve isolates the heat exchanger and heat transfer is inhibited. It is important to note that the effectiveness calculations are meaningless when the valve is bypassed as although exhaust gases are flowing through the heat exchanger device, they are not flowing over the heat exchanging surface.

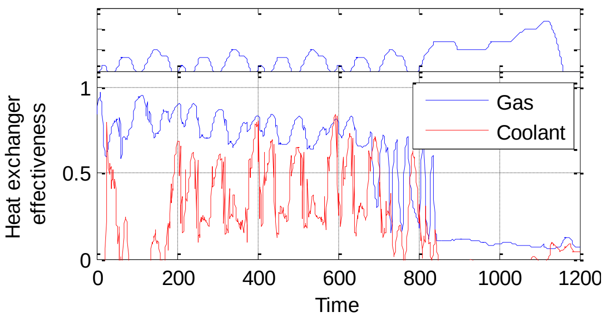

Figure 7: Exhaust gas heat exchanger effectiveness calculated from gas and coolant temperatures. NEDC vehicle speed trace shown as reference

Figure 8 shows the measured heat transfer across the exhaust gas heat exchanger for varying flow rates. There appears to be little dependency of heat transfer on coolant flow rate however there are large variations with respect to engine operating point and hence exhaust gas flow. From a theoretical perspective this would be expected as the heat transfer coefficient in the heat exchanger would be dominated by the gas side convection, meaning changes to the coolant flow would cause little difference compared to changes in exhaust gas flow. In total, approximately 500kJ was available over phase 1 and 1MJ over the complete NEDC.

4.2 Systems level behaviour

With the secondary circuit installation, the coolant and oil temperatures at the inlet to the oil cooler are compared in figure 9. The temperatures are similar; however the coolant remained colder than the oil throughout the warm-up. The second law of thermodynamics dictates that heat transfer from coolant to oil is impossible, and in fact heat transfer will occur in the opposite direction, effectively cooling the oil during warm-up. In practice the temperature difference is quite small and the resulting heat transfer will also be small, but clearly this is not the intended behaviour of the system. In this configuration, although heat is available from the exhaust gases, no use can be made of it in terms of oil warm-up.

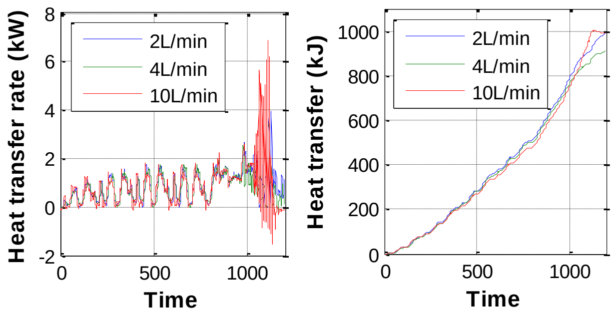

Figure 8: Heat transfer and heat flux across exhaust gas heat exchanger for varying coolant flow rates

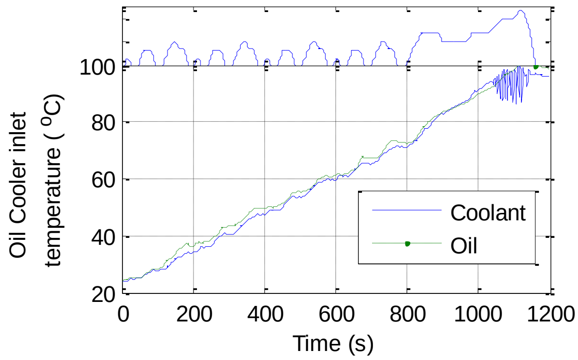

Figure 9: Coolant and oil temperatures at oil cooler inlet for secondary circuit configuration

The added flexibility of the integrated configuration was subsequently investigated to assess potential benefits from combined operation of the heat exchanger and the active thermal management system. In this configuration, some improvements of about 1-2°C in terms of engine warm-up were measured as shown in figure 10. This compares two separate hardware configurations whereby the heat exchanger and additional hoses were either installed or removed. This was not simply the difference between a change in exhaust gas heat exchanger butterfly valve position.

The behaviour of the system to different control settings is shown through the response models. The following results focus on phase 1 of the NEDC which represent the first 780s and roughly corresponds to the coolant warm-up. Figure 11 shows variations in cylinder liner temperature approximately half way down the bore. Figure 12 shows the same results for oil temperature. For both locations in the engine, the warm-up rate can be improved by increasing the coolant flow in the EGR cooler/ oil cooler/ exhaust heat exchanger leg. The other two control functionalities (EGR cooler type and oil cooler control valve) have opposing impacts on these two temperatures: these effectively control a trade-off between cylinder liner and oil warm-up.

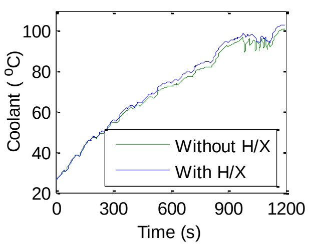

Figure 10: Coolant and oil temperatures with and without heat exchanger heat exchanger hardware installed

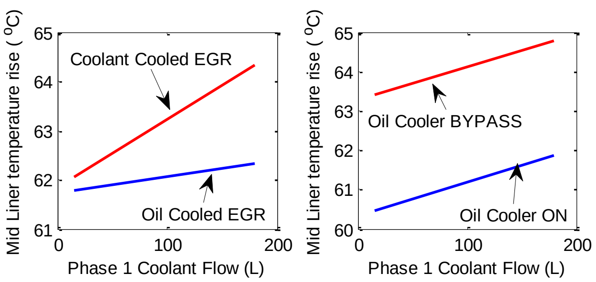

Figure 11: Variation in liner temperature warm-up with coolant flow, oil cooler and EGR cooler type

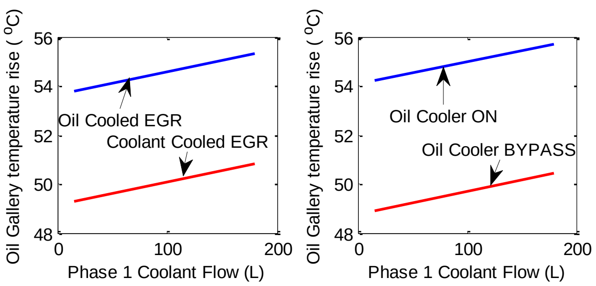

Figure 12: Variation of oil temperature warm-up with coolant flow, oil cooler and EGR cooler type

The trade-off in coolant and oil warm-up is shown in figure 13. This has been plotted for phase 1 of the drive cycle only and the temperature rises represent the change in temperature from cold-start. This shows that under some control conditions coolant warm-up can be preferred over oil and visa-versa. The results for the active thermal management system alone (ATM) are from previous work on this engine [2]. This trade-off is not new for the configuration with the heat exchanger, however two points are apparent:

1. The potential for oil warm-up with detriment to coolant warm-up is improved
2. The Pareto front is increased by about 1-4°C with the heat exchanger, demonstrating a small improvement in warm-up.

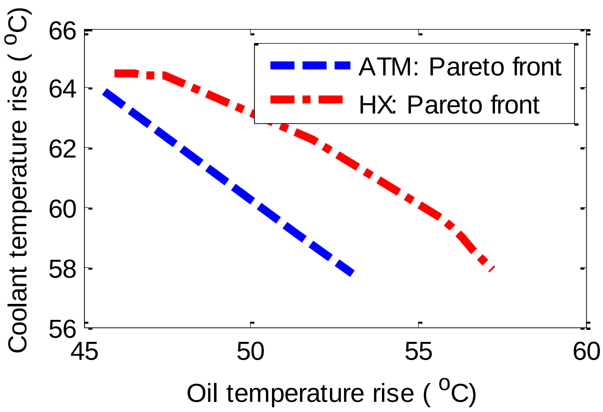

Figure 13: Coolant and oil warm-up trade-off for active thermal management without heat exchanger (ATM) and with heat exchanger (HX)

Analysing the warm-up rate is interesting, however ultimately it is benefits in engine performance that are sought, therefore modelling of emissions and fuel consumption was also performed. However, the variations in fuel consumption were so small that they could not be captured in the response model structure. The fuel consumption measurements from the configuration with exhaust gas heat exchanger were compared to previous experiments where this hardware was not installed [3]: these are shown in the form of bar charts for both phase 1 and the complete NEDC in figure 14. In these figures the bars do not represent experimental repeatability but rather the spread from respective DoE test plans. As a result, it is not the scatter that is of interest, but rather that all experiments with the exhaust gas heat exchanger give similar fuel consumption to those from the active thermal management system alone. It is clear from the fuel consumption results that there is no significant benefit from the exhaust gas heat exchanger.

Figure 15 shows the results from the NOx emissions response model which was fitted to the experimental data. This shows that the EGR cooler type has a significant effect on NOx emissions with oil-cooled EGR giving up to 10% benefit over coolant-cooled EGR. This has previously been shown to be a result of improved EGR gas cooling which reduced the intake gas temperature. The interactions with the coolant flow and oil cooler setting appear to be a result of higher oil temperatures impacting on the EGR gas cooling effectiveness.

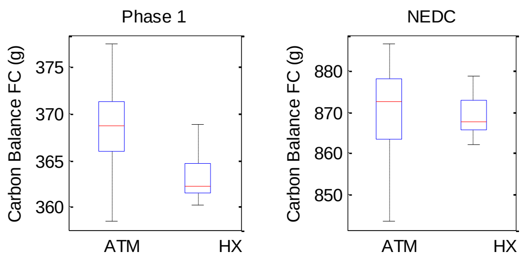

Figure 14: Fuel consumption results for active thermal management system without exhaust gas heat exchanger (ATM) and with exhaust gas heat exchanger (HX)

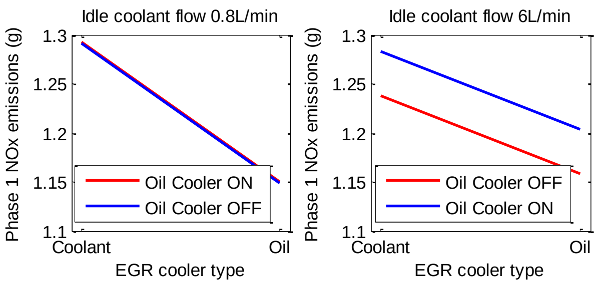

Figure 15: Variation of NOx emissions with coolant flow, oil cooler and EGR cooler type

4.3 Analytical results

Figure 16 shows the analytical results for both simulation approaches. The simulations show that to achieve a significant benefit, the inertia of the additional system must be low. Figure 16 (a) shows that to achieve a faster coolant warm-up compared to that of oil, the additional inertia should be less than 3kg water. This first approach ignores the impact of heat transfer from coolant to oil. Figure 16 (b) shows that when this heat transfer is included, to have any significant effect and provide significant heat transfer to oil, ideally the inertia should be even less. During these experiments it is estimated that the equivalent thermal inertia of the exhaust gas heat exchanger system was approximately 3kg water.

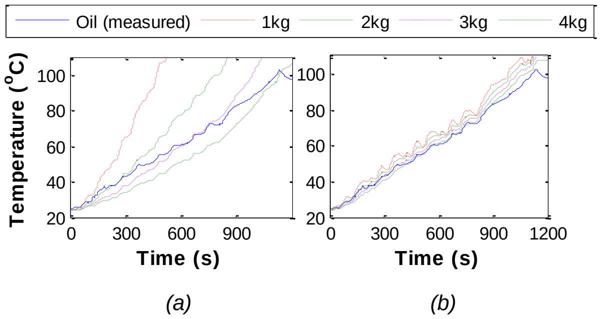

Figure 16: Coolant temperature for (a) *heat-to-coolant* and (b) *heat-to-coolant-to-oil* simulations for different thermal inertias of the additional heat exchanger system (thermal inertia expressed as equivalent water mass)

5. **Discussion**

In this work an exhaust gas heat exchanger was installed into a prototype thermal management system. The aim was to use otherwise waste heat to improve engine warm-up

The heat exchanger itself had a high effectiveness, although heat transfer was dominated by changes in exhaust flow. This means the benefit of the device is strongly dependent on duty cycle. Over the NEDC, 1MJ of heat was extracted, although half of this is only available in phase 2 when warm-up is less critical. The true benefits will arise from early availability of heat.

The active thermal management system offers a warm-up trade-off between coolant and oil temperature using the flow control valves. The inclusion of the exhaust gas heat exchanger allows this trade-off to be extended for oil temperature rise. This shows that there is potential to increase oil warm-up rate at the detriment of coolant warm-up should this be beneficial to engine fuel consumption. There was also an improvement in the warm-up trade-off Pareto front showing that the heat exchanger is contributing to overall warm-up. Fuel consumption results showed no improvement despite this faster warm-up meaning that the benefits are too small to cause any performance benefits.

In practice, the main limitation for the system stems from requirements of exhaust gas after-treatment devices. With current and foreseeable legislation and technology development trends, it is difficult to see an exhaust gas heat exchanger taking priority over these devices. This impacts both the available exhaust gas temperature and the increased thermal inertia required to integrate the device in the cooling circuit. In this investigation it is the latter that rendered the impact of the new system on fuel consumption insignificant. Although 500kJ of heat was available over the first phase of the drive cycle, this was roughly equivalent to the energy required to heat the additional thermal mass at the same rate as the engine. Future design work should concentrate on minimising this thermal mass

A benefit in NOx emissions was apparent when using oil-cooled EGR. This benefit is due to improved EGR gas cooling from the oil system compared to the coolant system. The reasons for this have not yet been determined, however they could be a result of higher oil flow rates; colder oil temperature or the slightly longer gas path route for the oil EGR cooler, all compared to the coolant system.

The sensitivity of the system to engine duty cycle show that real world benefits may not be reflected in this study. In addition, the system may provide more significant benefits at colder start temperatures. In this study 25^{o}C has been used as it is the standard for vehicle homologation, however performance may be different under colder conditions.

Insulation of the exhaust pipe was not considered in this work, however this may become important in getting heat to the exhaust components. The limitations will be whether heat is rejected sufficiently without damage to the after-treatment devices at fully warm operating temperatures.

6. Conclusions

An exhaust gas heat exchanger was installed into a prototype thermal management system with the aim of improving engine warm-up and fuel consumption. The following conclusions were drawn from this exercise:

1. The heat exchanger was effective at extracting heat from exhaust gases, although the available heat was heavily dependent on duty cycle. 500kJ of heat was available during phase 1 of the NEDC.
2. The thermal mass of the additional system should be reduced to at least that of 2kg water and ideally less than 1kg.
3. A warm-up trade-off from the active thermal management system was apparent and the heat exchanger appeared to provide a 1-4^{o}C improvement in warm-up, although this was not reflected in fuel consumption benefits.
4. Performance of the system is strongly linked to duty cycle. The experiments in this work may not make best use of the heat exchanger and on vehicle applications cabin heating and other powertrain elements may show a more viable application

7. Acknowledgements

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9. Glossary

ATM: Active Thermal Management

DoE: Design of Experiments

DPF: Diesel Particulate Filter

EGR: Exhaust Gas Recirculation

FC: Fuel Consumption

HX: Heat Exchanger

NEDC: New European Drive Cycle

NOx: Oxides of Nitrogen

PRT: Pressure Regulated Thermostat

$(hA)_{oilcooler}$: Oil cooler heat transfer coefficient (kJ/K)

C_h: Hot Fluid Heat Capacity (kJ/kgK)

C_i: Heat capacity of coolant circuit component (Kj/kgK)

C_{min}: Minimum Heat Capacity of two Fluids (kJ/kgK)

dT_{cool}: Coolant temperature change (^oC)

m_i: mass of coolant circuit component (kg)

q: Actual Heat transfer (kJ)

q_{max}: Maximum possible Heat Transfer (kJ)

$T_{c,i}$: Cold Fluid Inlet Temperature (^oC)

T_{cool}: Coolant temperature (^oC)

$T_{h,i}$: Hot Fluid Inlet temperature (^oC)

$T_{h,o}$: Hot Fluid Outlet Temperature (^oC)

T_{oil}: Oil temperature (^oC)

ε: Heat exchanger effectiveness